

AD-A102 792

# TECHNICAL LIBRARY

AD A-102 792

TECHNICAL REPORT ARLCB-TR-81024

## A COMPARISON OF THREE POINT AND FOUR POINT LOADING IN ELASTIC-PLASTIC BENDING OF BEAMS

R. Vincent Milligan

June 1981



US ARMY ARMAMENT RESEARCH AND DEVELOPMENT COMMAND  
LARGE CALIBER WEAPON SYSTEMS LABORATORY  
BENÉT WEAPONS LABORATORY  
WATERVLIET, N. Y. 12189

AMCMS No. 612105H840011

DA Project No. 1L162105AH84

PRON No. 1A0217131A1A

APPROVED FOR PUBLIC RELEASE; DISTRIBUTION UNLIMITED

#### DISCLAIMER

The findings in this report are not to be construed as an official Department of the Army position unless so designated by other authorized documents.

The use of trade name(s) and/or manufacturer(s) does not constitute an official indorsement or approval.

#### DISPOSITION

Destroy this report when it is no longer needed. Do not return it to the originator.

REPORT DOCUMENTATION PAGE		READ INSTRUCTIONS BEFORE COMPLETING FORM
1. REPORT NUMBER ARLCB-TR-81024	2. GOVT ACCESSION NO.	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle) A COMPARISON OF THREE POINT AND FOUR POINT LOADING IN ELASTIC-PLASTIC BENDING OF BEAMS		5. TYPE OF REPORT & PERIOD COVERED Final
7. AUTHOR(s) R. Vincent Milligan		6. PERFORMING ORG. REPORT NUMBER
9. PERFORMING ORGANIZATION NAME AND ADDRESS US Army Armament Research & Development Command Benet Weapons Laboratory, DRDAR-LCB-TL Watervliet, NY 12189		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS AMCMS No. 612105H840011 DA Project No. 1L162105AH84 PRON No. 1A0217131A1A
11. CONTROLLING OFFICE NAME AND ADDRESS US Army Armament Research & Development Command Large Caliber Weapon Systems Laboratory Dover, NJ 07801		12. REPORT DATE June 1981
14. MONITORING AGENCY NAME & ADDRESS (if different from Controlling Office)		13. NUMBER OF PAGES 22
15. SECURITY CLASS. (of this report) UNCLASSIFIED		
15a. DECLASSIFICATION/DOWNGRADING SCHEDULE		
16. DISTRIBUTION STATEMENT (of this Report) Approved for public release; distribution unlimited.		
17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)		
18. SUPPLEMENTARY NOTES		
19. KEY WORDS (Continue on reverse side if necessary and identify by block number) Beams Straightening Bending Elastic-Plastic Analysis		
20. ABSTRACT (Continue on reverse side if necessary and identify by block number) A comparison between the maximum deflections resulting from beams symmetrically loaded into the elastic-plastic region is presented for the case of three and four point bending. The results indicate that significantly larger deflections can be obtained for the case of four point bending while keeping the maximum fiber strains approximately the same. It appears that using the four point bending approach holds certain advantages in straightening operations for the (CONT'D ON REVERSE)		

20. ABSTRACT (CONT'D)

removal of permanent deflections with the possible elimination of hot straightening in some instances. The concept of distribution of plastic deformation is put forth as a possible explanation of the advantage of one method over the other.

## TABLE OF CONTENTS

	<u>Page</u>
INTRODUCTION	1
RESULTS AND DISCUSSION	6
CONCLUSIONS	9
REFERENCES	11
APPENDIX A	A-1
APPENDIX B	B-1

## ILLUSTRATIONS

1. Load and Moment Diagrams for Three Point Bending.	2
2. Load and Moment Diagrams for Four Point Bending.	4
3. Load vs. Deflection.	12
4. Moment of Inertia vs. Deflection.	13
5. Moment of Inertia vs. Deflection.	14
6. Ratio of Four Point Deflection to Three Point Deflection.	15
7. Ratio of Four Point Deflection to Three Point Deflection vs. Outside Radius.	16
8. Maximum Deflection Under Load vs. Length of Plastic Zone.	17
A1. Strain vs. Depth of Elastic-Plastic Interface.	A-1
B2. Strain vs. Depth of Cross-Section.	B-1

## INTRODUCTION

One of the questions that arises in the area of straightening components deals with the magnitude of deflections and the corresponding strains which could adversely affect material properties and limit subsequent usage of the component. An equally pertinent question centers on the need for hot straightening to eliminate excessive permanent deflections or bows in components due to processing operations. With these questions forming a backdrop, the writer performed an analysis comparing three and four point bending from the standpoint of seeing whether larger permanent deflections could be removed using the four point method and thus possibly obviating the need for hot straightening. It was hoped that this could be done without causing additional material degradation as a result of straining the material into the plastic region.

The intent of this report is to present the results of a theoretical elastic-plastic analysis as a means of comparing the two methods.

The theory used as a basis for the calculations in this report is essentially the same as developed by Seely and Smith.<sup>1</sup> We first consider the case for a concentrated load at midspan as shown in Figure 1(a). Using the dummy load method, which is practically the same as the principle of virtual work, the deflection can be calculated from the integral

$$\delta = \int m d\alpha \quad (1)$$

---

<sup>1</sup>F. B. Seely and J. O. Smith, Advanced Mechanics of Materials, 2nd Ed., Wiley and Sons, 1952.

where  $m$  is the moment due to the dummy load and  $d\alpha$  the relative rotation of one plane of a cross section with respect to another a distance  $dx$  apart. For the linear region  $d\alpha$  is equal to  $(M/EI) dx$ , where  $M$  is the moment due to the real load,  $E$  is the modulus of elasticity, and  $I$  is the moment of Inertia. We first integrate through the elastic region from 0 to  $\lambda$ , where  $\lambda$  is the distance from the left support to the end of the elastic region. We then have

$$\delta_1 = \int_0^\lambda \frac{M}{EI} dx \quad (2)$$

The limit  $\lambda$  depends on the loading conditions and can be obtained by ratio from the moment diagram. Referring to Figure 1(b)

$$\frac{\lambda}{M_y} = \frac{\lambda/2}{M_{\max}} \quad (3)$$

hence

$$\lambda = \frac{M_y}{M_{\max}} (\lambda/2) \quad (4)$$

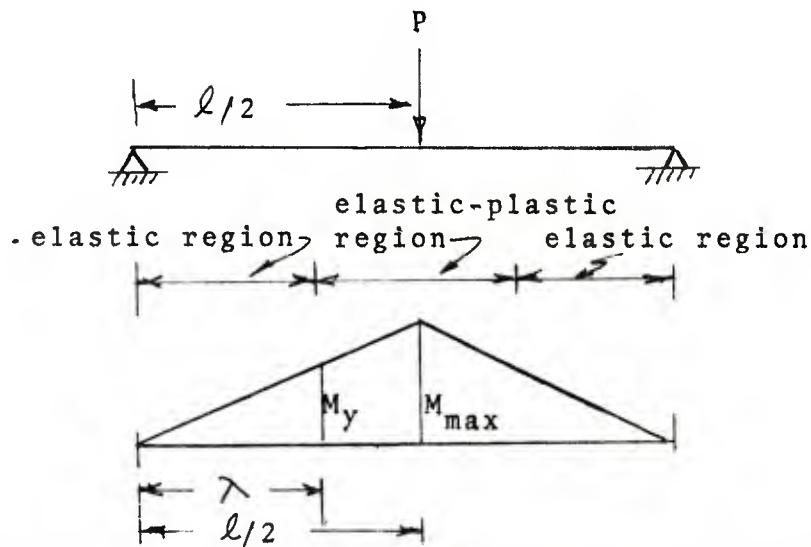


Figure 1. Load and Moment Diagrams for Three Point Bending.

As the moment increases from the yield moment  $M_y$  to the fully plastic moment  $M_{fp}$ ,  $\lambda$  decreases, indicating that the length of the elastic-plastic zone increases.

We next integrate through the elastic-plastic zone from  $\lambda$  to  $\lambda/2$ . For the elastic-plastic region  $d\alpha$  now takes the form of

$$d\alpha = \frac{M_y}{EI} \frac{n^{1/2} dx}{\sqrt{K - \frac{M_p}{M_y}}} \quad (5)$$

One can refer to the Appendix for the development of this expression. Hence,

$$\delta_2 = \int_{\lambda}^{\lambda/2} \frac{M_y}{EI} \frac{n^{1/2} mdx}{\sqrt{K - \frac{M_p}{M_y}}} \quad (6)$$

The total deflection can then be obtained by adding deflections  $\delta_1$  and  $\delta_2$  and multiplying by two since we have symmetrical loading.

For a simply supported beam having a concentrated load,  $M = Px/2$  and  $m = x/2$  for the region  $0 < x < \lambda/2$ . Hence:

$$\delta_1 = \int_0^{\lambda} \frac{P}{4} \frac{x^2}{EI} dx \quad (7)$$

$$\text{and } \delta_2 = \frac{M_y}{EI} n^{1/2} \int_{\lambda}^{\lambda/2} \frac{x}{2} \frac{1}{\sqrt{K - \frac{Px}{2M_y}}} dx \quad (8)$$

Inserting the factor two previously mentioned, we obtain:

$$\delta = \delta_1 + \delta_2 = \frac{P}{2EI} \int_0^{\lambda} x^2 dx + \frac{M_y}{EI} n^{1/2} \int_{\lambda}^{\lambda/2} \frac{x}{\sqrt{K - \frac{Px}{2M_y}}} dx \quad (9)$$

Integrating and inserting limits, we get the equivalent expression given by

Seely and Smith<sup>1</sup>

$$\delta = \frac{P\ell^3}{48EI} \frac{M_y^3}{M_{\max}^3} \left[ 1 + 2n(2K+1) - 2n^{1/2} \left( 2K + \frac{P\ell}{4M_y} \right) \left( K - \frac{P\ell}{4M_y} \right)^{1/2} \right] \quad (10)$$

It is readily apparent that this expression degenerates to the elastic equation when  $M_{\max} = M_y$ . That is,

$$\delta = \frac{P\ell^3}{48EI} \quad (11)$$

A computer program was written for equation (10) and given the acronym DEFL. The program has a Do Loop on P. The results shown graphically were obtained from this program.

We next consider the case for two concentrated loads symmetrically located relative to the midspan as shown in Figure 2(a).

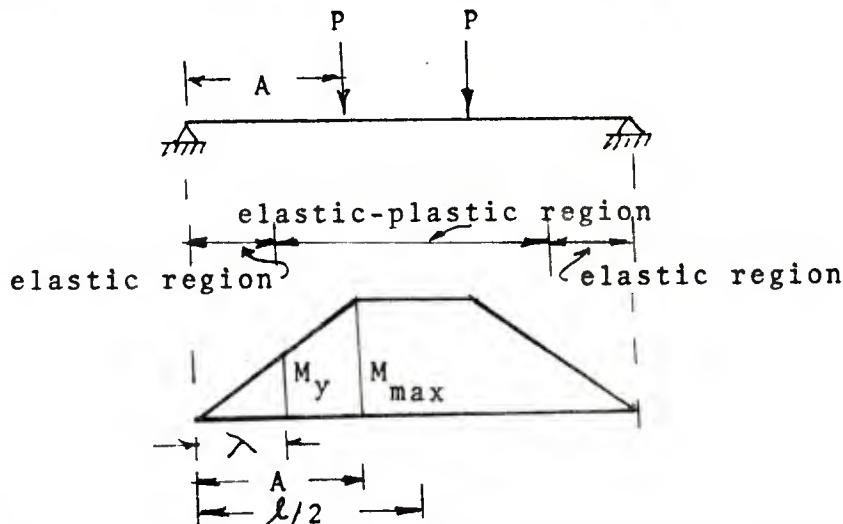


Figure 2. Load and Moment Diagrams for Four Point Bending.

<sup>1</sup>F. B. Seely and J. O. Smith, Advanced Mechanics of Materials, 2nd Ed., Wiley and Sons, 1952.

In this case, we have to integrate through three regions, 0 to  $\lambda$ ,  $\lambda$  to A, and A to  $\lambda/2$ .

$$\delta_1 = \int_0^\lambda \frac{M_m dx}{EI} = \frac{1}{EI} \int_0^\lambda P_x \left(\frac{x}{2}\right) dx \quad (12)$$

where the limit  $\lambda$  by ratio is equal to  $\frac{M_y}{M_{max}} A$  as can be noted from Figure 2(b).

$$\delta_2 = \int_\lambda^A \frac{M_y}{EI} \frac{n^{1/2} mdx}{\sqrt{K - \frac{M_p}{M_y}}} = \frac{M_y}{EI} n^{1/2} \int_\lambda^A \frac{x/2}{\sqrt{K - \frac{P_x}{M_y}}} dx \quad (13)$$

and

$$\delta_3 = \int_A^{\lambda/2} \frac{M_y}{EI} \frac{n^{1/2} mdx}{\sqrt{K - \frac{M_{max}}{M_y}}} = \frac{M_y n^{1/2}}{EI} \frac{1}{\sqrt{K - \frac{M_{max}}{M_y}}} \int_A^{\lambda/2} \left(\frac{x}{2}\right) dx \quad (14)$$

Evaluating these integrals, and inserting the factor of two because of symmetry and integrating over half the length, the maximum deflection at  $\lambda/2$  becomes

$$\delta = \delta_1 + \delta_2 + \delta_3 \quad (15)$$

$$\delta = \frac{PA^3}{3EI} \frac{M_y^3}{M_{max}^3} + \frac{2}{3} \frac{M_y^3}{EI} \frac{n^{1/2}}{P^2} \left[ \left( -2K + \frac{PA}{M_y} \right) \left( K - \frac{PA}{M_y} \right)^{1/2} + (2K+1)(K-1)^{1/2} \right] + \frac{M_y}{EI} \frac{n^{1/2}}{\sqrt{K - \frac{PA}{M_y}}} \left[ \frac{\lambda^2}{8} - \frac{A^2}{2} \right] \quad (16)$$

A computer program was written for equation (16) and given the acronym SSDEFL2P. The program has a Do Loop on P. The results shown graphically were obtained from this program.

If we specialize this equation for  $A = \ell/2$ , the second and third terms drop out and we get

$$\delta = \frac{P\ell^3}{24EI} \quad (17)$$

This differs from equation (11) by a factor of two since we have two loads acting at  $\ell/2$ .

#### RESULTS AND DISCUSSIONS

As a means of comparing three point bending with four point bending, the following cases were calculated using the previously mentioned computer programs with the following input data:

$$R_o = 3.0, 3.5, 4.0, 4.5 \text{ inches}$$

$$R_i = 2.0 \text{ inches}$$

$$E = 30 \times 10^6 \text{ psi}$$

$$\sigma_y = 160 \text{ Ksi}$$

$$\ell = 60 \text{ inches}$$

$$A = 15, 20, 25 \text{ inches}$$

The load  $P$  was located at  $\ell/2$  for the case of three point bending. This gave a total of 16 load-deflection curves. An example for the case of  $R_o = 3.0$  inches with  $A = 15, 20$ , and  $25$  inches is shown in Figure 3. Assuming linear elastic unloading, the permanent deflections were obtained by projecting downward from the maximum deflection at maximum load, to the deflection axis with a line parallel to the initial elastic loading line. It should be pointed out that load  $P$  which would be applied to the loading fixture to accomplish the four point loading is actually equal to  $2P$  and is consequently twice the value of  $P$  applied in the three point case. The same is also true for all

subsequent diagrams and figures comparing three point and four point loading. From these 16 curves we constructed two others, Figures 4 and 5. Figure 4 is a plot of moment of inertia ( $I$ ) vs. deflection for the three point loading and three different values of  $A$  for the four point loading. The values of  $I$  for four different outside radii are marked. The solid lines show the maximum deflection under load and the dashed lines are for the permanent deflections. The plasticity condition is such that the elastic-plastic interface has moved a distance equal to 75 percent of the outside radius from the outside fiber toward the neutral axis. Equation 35 of reference 2 gives a relationship between the outside radius  $R_o$ , the distance from the neutral axis to the elastic-plastic interface  $\rho$ , and the yield strain,  $\epsilon_y$ . That is,

$$\epsilon_{\max} = \frac{R_o}{R_o - \rho} \epsilon_y$$

For the 75 percent plasticity condition used for comparison purposes for the two methods,  $R_o$ ,  $\rho$ , and  $\epsilon_y$  would be the same, therefore the maximum strains would be the same for both bending methods. The same reasoning would hold for the nearly 100 percent plasticity condition also considered.

Figure 5 is the same plot as Figure 4 except the plasticity condition is such that the elastic-plastic interface has moved a distance equal to nearly 100 percent of the outside radius. Table I gives a summary of the various geometrical parameters, loads, moments, and deflections.

---

<sup>2</sup>R. V. Milligan, "Moment-Strain Relationships in Elastic-Plastic Bending of Beams," to be published.

Figure 6 was obtained by taking ratios of maximum deflections under load for the four point case to the three point case for a constant  $R$ . For the 100 percent plasticity condition, the maximum ratio is 7.85 for  $A = 15$  and  $R_0 = 3.0$  while the minimum ratio is 3.25 for  $A = 25$  and  $R_0 = 4.5$ . For the 75 percent plasticity condition, the maximum ratio is 2.90 for  $A = 15$  and  $R_0 = 3.0$  while the minimum ratio is 1.61 for  $A = 25$  and  $R_0 = 4.5$ . It is thus evident for both plasticity conditions that the maximum ratios occur for the beam having a smaller cross section and as the loads get nearer to the supports.

Figure 7 is the same as Figure 6, except the deflection ratios are permanent deflections. Again the maximum ratio for the 100 percent plasticity condition occurs for  $A = 15$  and  $R_0 = 3.0$  while the minimum ratio occurs for  $A = 25$  and  $R_0 = 4.0$ . For the 75 percent plasticity condition, the maximum ratio occurs for  $A = 15$  at a value of  $R_0$  between 3.5 and 4.0.

In summary, the ratio of maximum deflections for the 100 percent plasticity condition run from a low of about 3.25 to a high of about 7.75. For the 75 percent plasticity condition they run from a low of about 1.75 to a high of about 3.75. The ratios are even greater for the permanent deflections. Here they run from a low of 6.25 to a high of 19 for the 100 percent condition and from about 2 to 8.75 for the 75 percent condition. It is thus very evident, based on the theory presented here, that four point bending can be a means of significantly increasing the deflections compared to three point bending, while at the same time keeping the strains and material degradation at the same level.

In an effort to propose a reason for the apparent advantage of four point bending, the author submits the concept of distributed plastic flow. Figure 8 is a plot of maximum deflection under load vs. length of plastic zone. The length of plastic zone can be determined by simple ratio as follows. For the four point case:

$$\frac{x}{M_y} = \frac{A}{M_{max}} \text{ hence } x = \frac{M_y}{P_{max}A} \cdot A = \frac{M_y}{P_{max}}$$

where  $s$  is the distance from the left support to the beginning of the plastic zone. For the three point case:

$$\frac{x}{M_y} = \frac{\ell/2}{M_{max}} \text{ hence } x = \frac{M_y}{M_{max}} (\ell/2) = \frac{M_y}{P_{max}/2}$$

Figure 8 shows sketches depicting the size of the elastic-plastic zones both for the three point case and three different values of  $A$  for the four point case. Obviously, as the loads move outward toward the supports, the elastic-plastic region becomes larger. The plot of  $\delta_{max}$  vs. length of plastic zone is nearly linear. This tends to support the concept that as one distributes the plastic flow one can increase the deflection without appreciably increasing the maximum fiber strain with its probable consequence of degradation.

#### CONCLUSIONS

From the analysis presented it appears that larger deflections under load can be obtained and consequently larger permanent deformations can be removed by using the four point bending method compared with three point bending while keeping the maximum fiber strains the same in both cases.

The concept of distributed plastic flow appears to be a valid reason for the advantage of the four point method relative to the three point method.

#### REFERENCES

1. F. B. Seely and J. O. Smith, Advanced Mechanics of Materials, 2nd Ed., Wiley and Sons, 1952.
2. R. V. Milligan, "Moment-Strain Relationships in Elastic-Plastic Bending of Beams," to be published.
3. J. H. Faupel, Engineering Design, Wiley and Sons, 1964, p. 395.

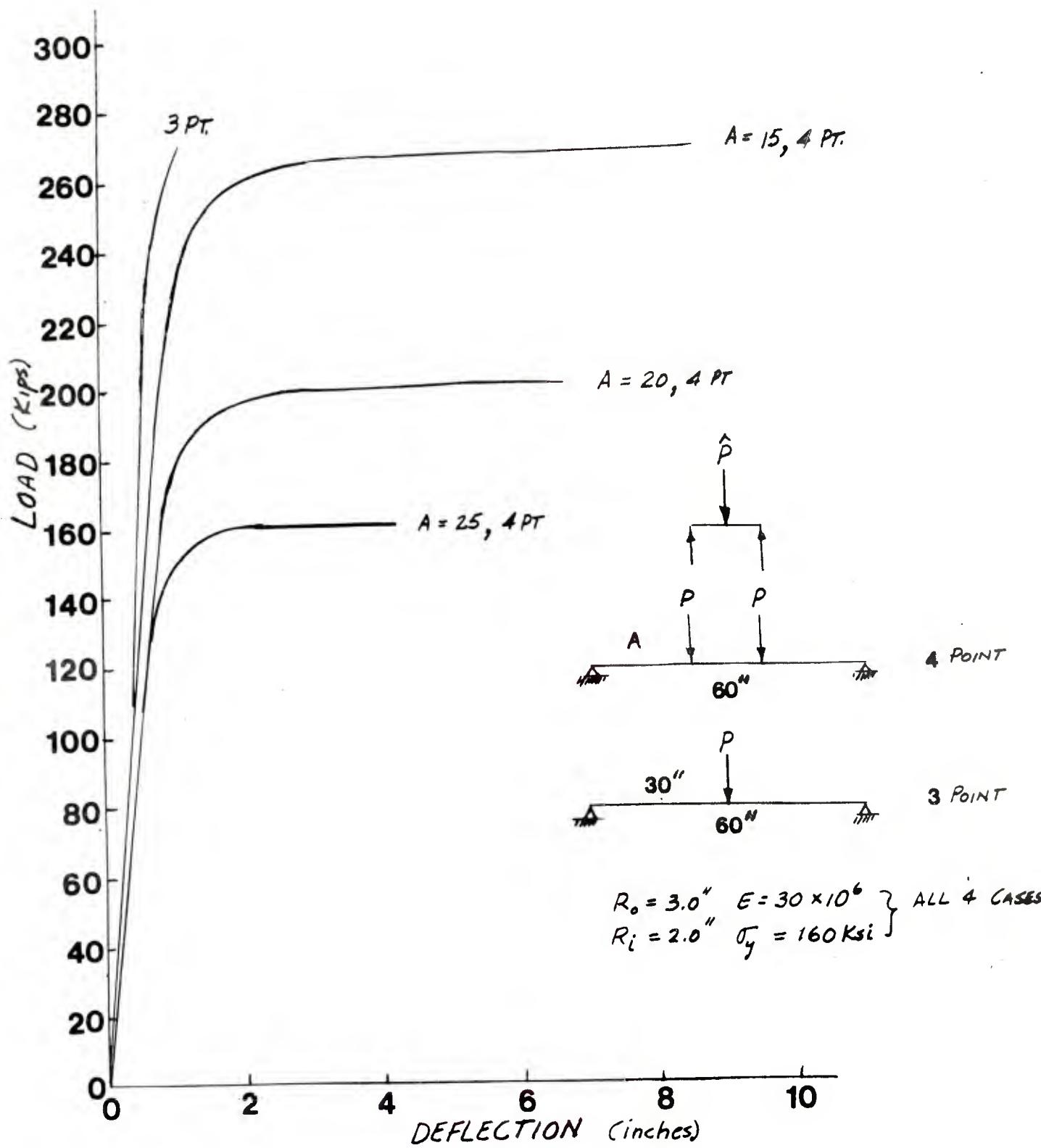


Figure 3. Load vs. Deflection.

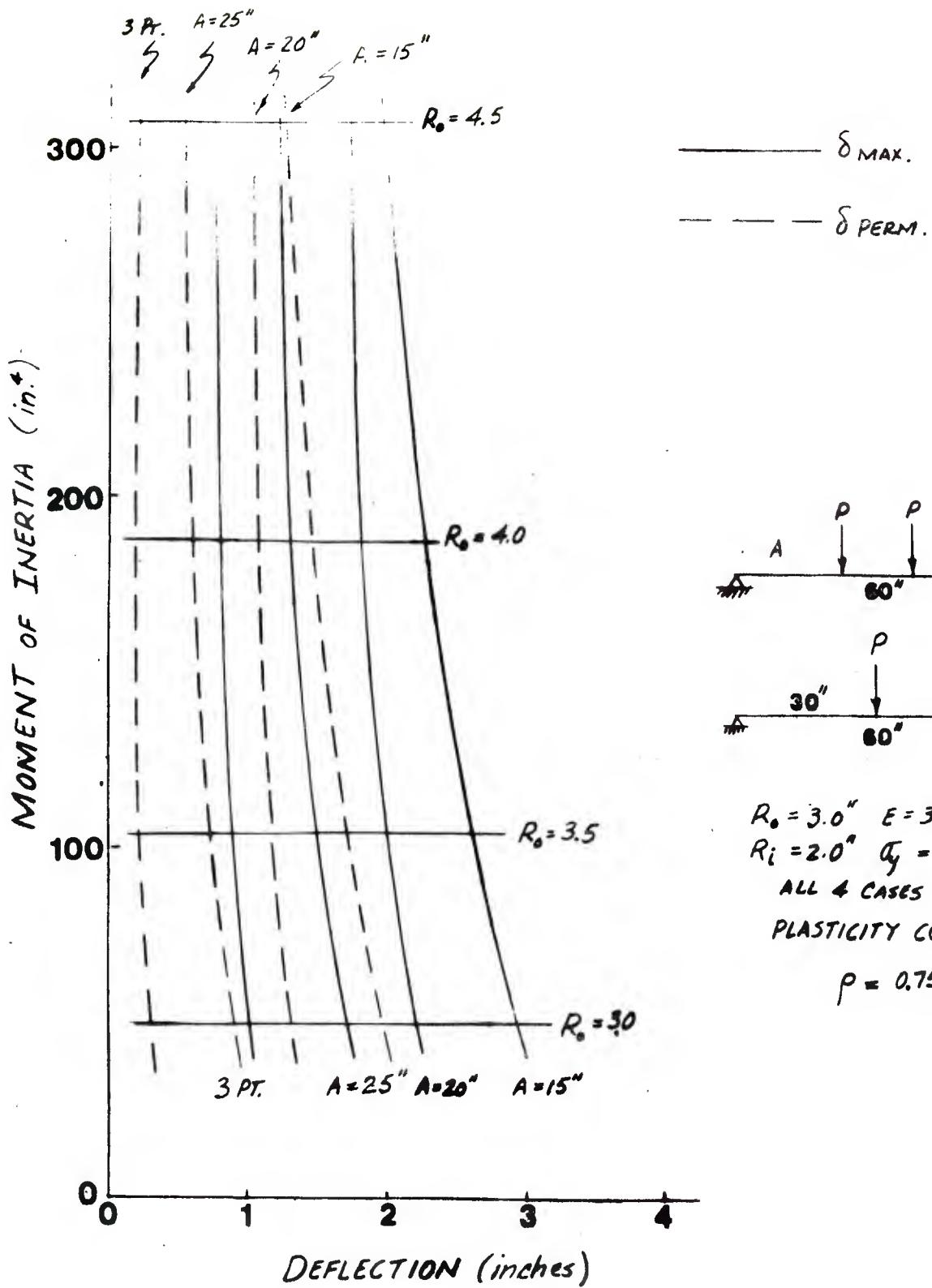


Figure 4. Moment of Inertia vs. Deflection.

MOMENT OF INERTIA (in.<sup>4</sup>)

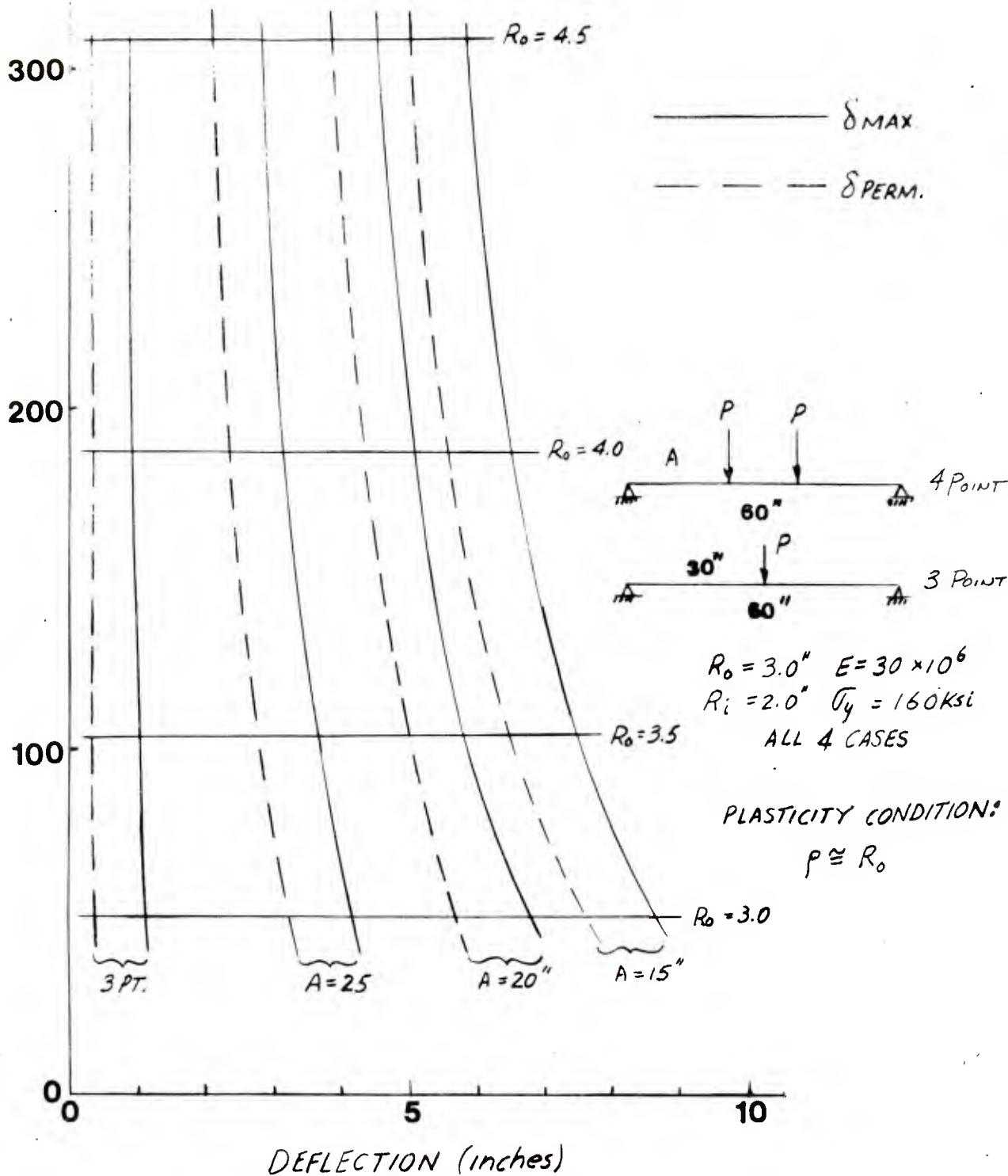


Figure 5. Moment of Inertia vs. Deflection

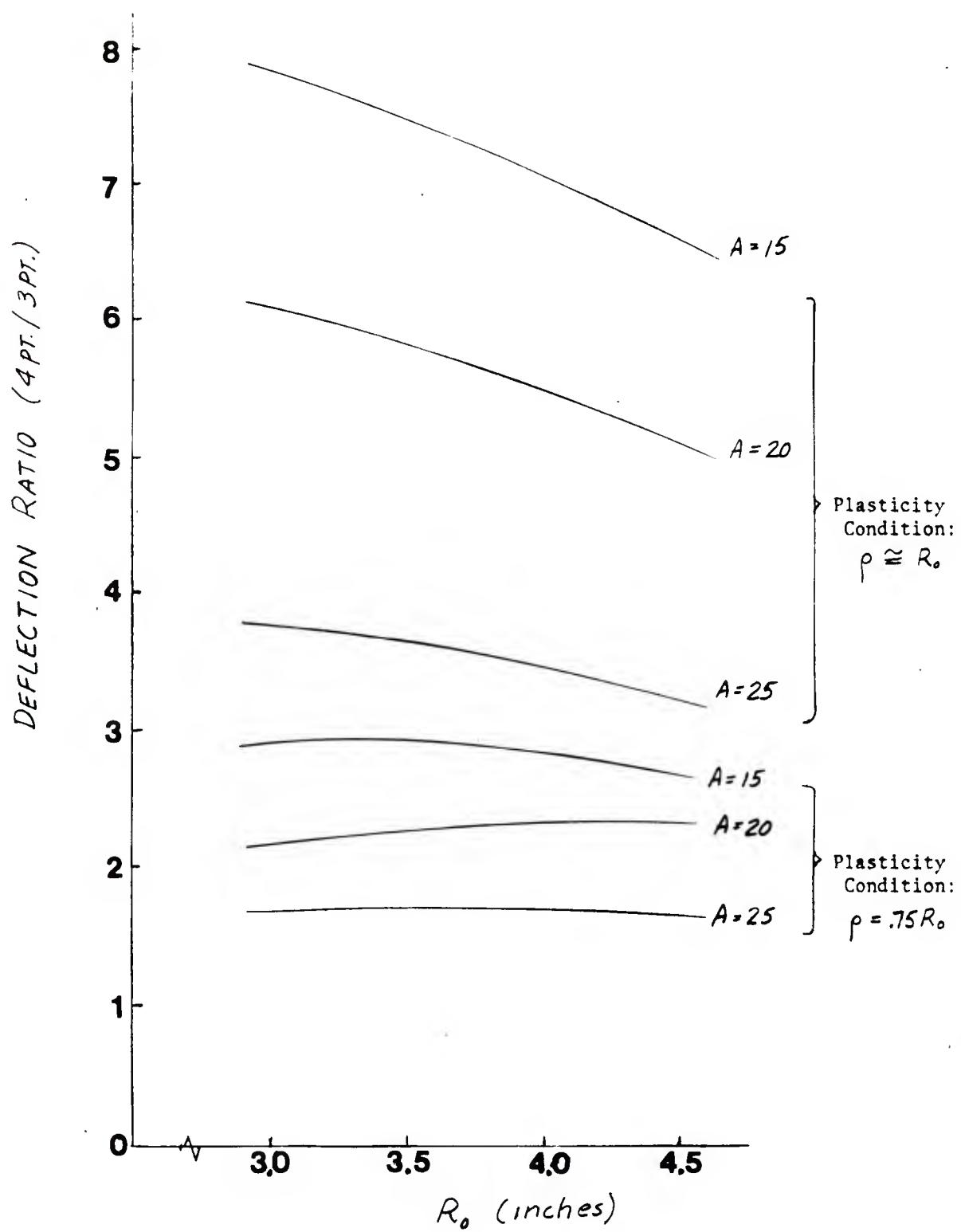


Figure 6. Ratio of Four Point Deflection to Three Point Deflection.

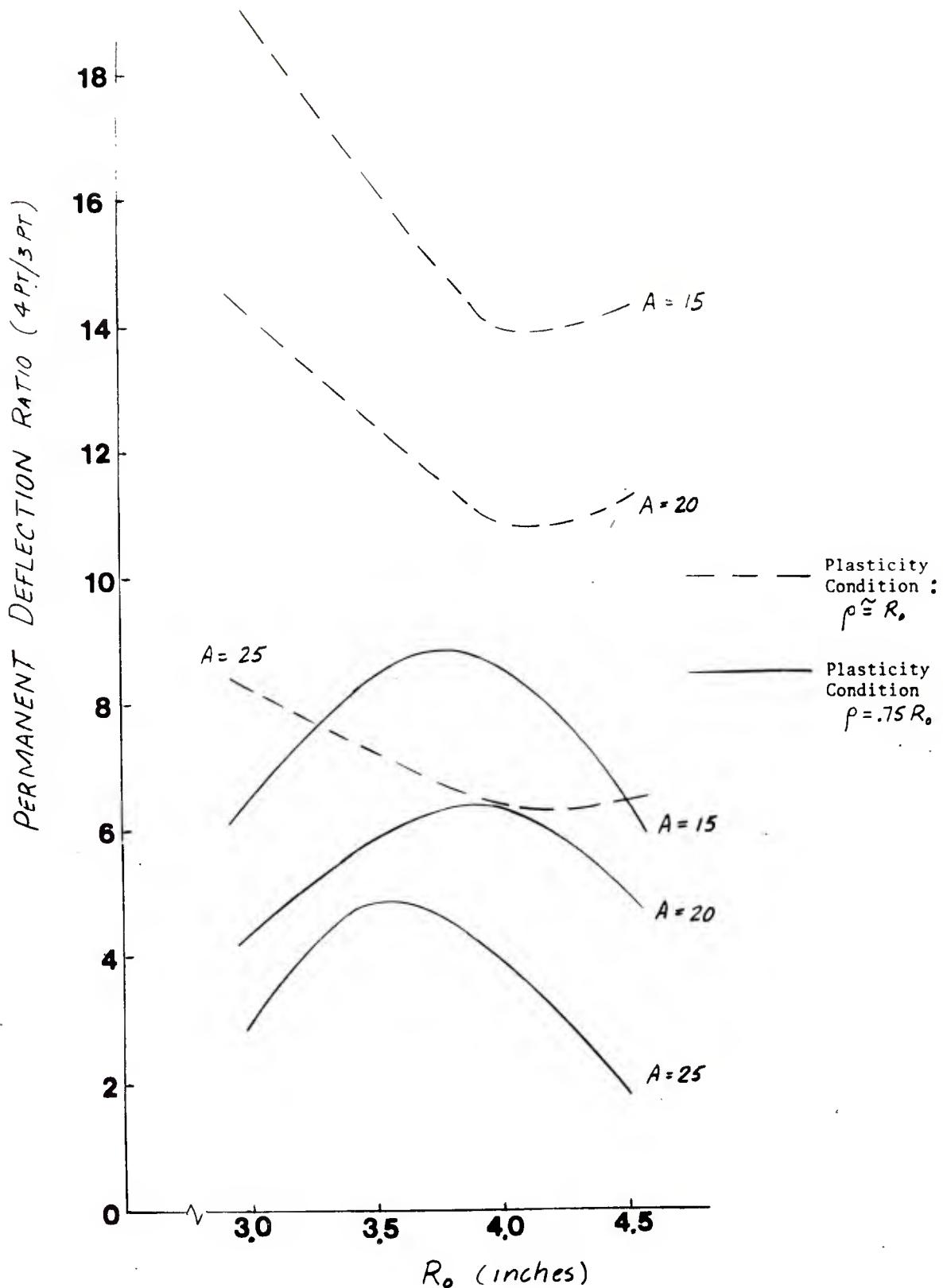


Figure 7. Ratio of Four Point Deflection to Three Point Deflection vs. Outside Radius.

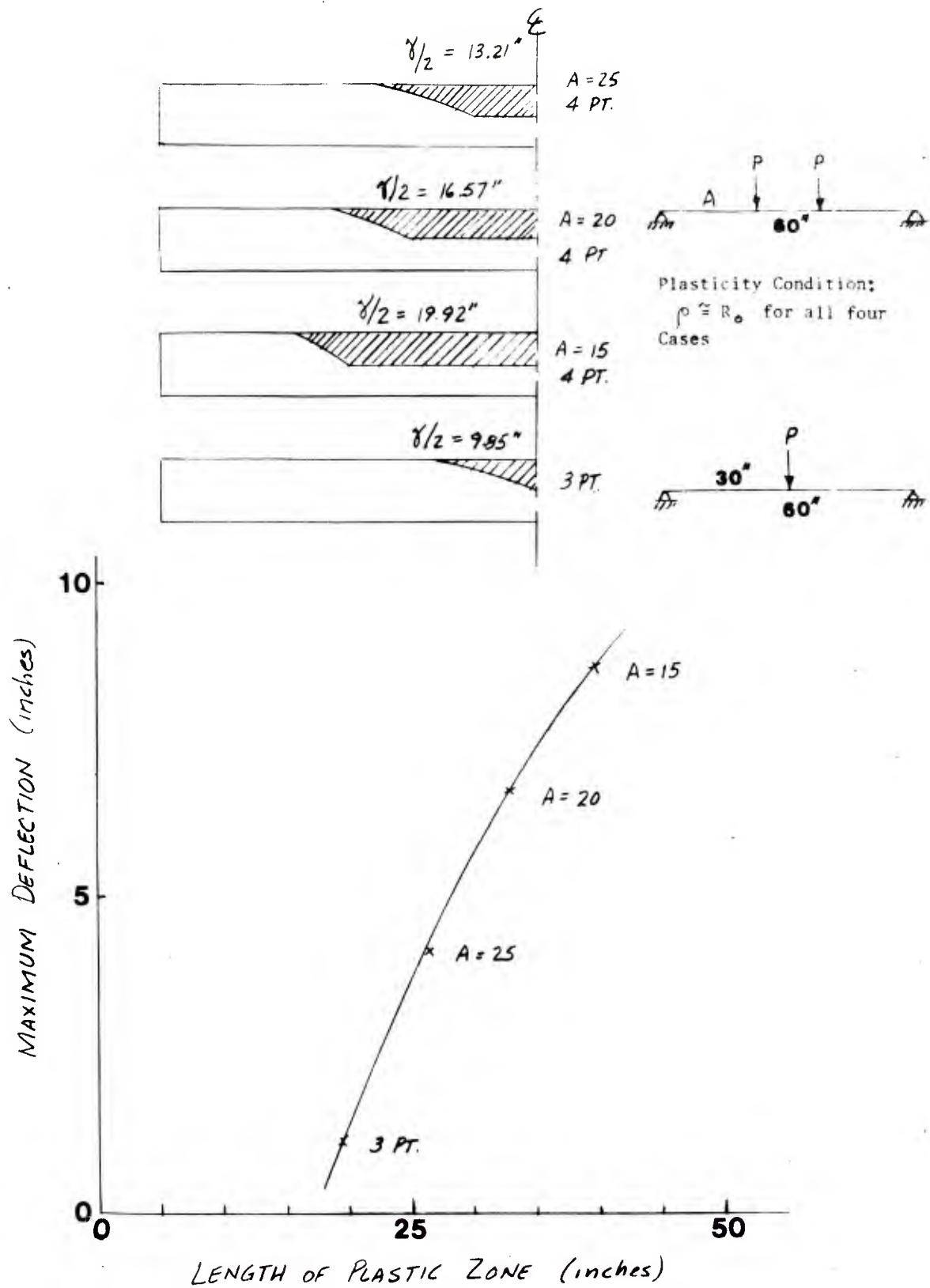


Figure 8. Maximum Deflection Under Load vs. Length of Plastic Zone.

## APPENDIX A

In a previous report<sup>2</sup> the bending moment corresponding to an elastic-plastic interface depth  $\rho$  for a rectangular beam was developed and shown to be

$$M_\rho = \frac{\sigma_y b}{6} \left[ h^2 + 2\rho h - 2\rho^2 \right] \quad (A1)$$

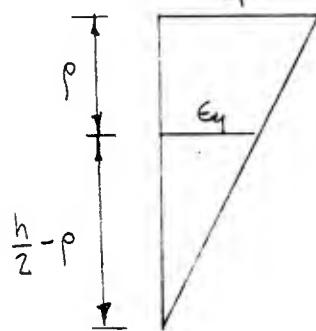


Figure A1. Strain vs. Depth of Elastic-Plastic Interface.

From Figure A1 we get the following relationship

$$\frac{\epsilon_\rho}{\epsilon_y} = \frac{1}{1 - \frac{2\rho}{h}} \quad (A2)$$

Solving for  $\rho$ , we obtain

$$\rho = \frac{h}{2} \left( 1 - \frac{\epsilon_y}{\epsilon_\rho} \right) \quad (A3)$$

Now substituting this into equation (A1), we get the bending moment corresponding to the elastic-plastic interface being at a depth equal to  $\rho$ .

$$M_\rho = \frac{\sigma_y b h^2}{6} \left[ \frac{3}{2} - \frac{1}{2} \frac{\epsilon_y^2}{\epsilon_\rho^2} \right] \quad (A4)$$

---

<sup>2</sup>R. V. Milligan, "Moment-Strain Relationships in Elastic-Plastic Bending of Beams," to be published.

ratio

$$\frac{M_p}{M_y} = \frac{3}{2} - \frac{1}{2} \frac{\epsilon_y^2}{\epsilon_p^2} \quad (A5)$$

Multiplying and dividing the second term of equation (A5) by  $(dx/\frac{h}{2})^2$  we get

$$\frac{M_p}{M_y} = \frac{3}{2} - \frac{1}{2} \frac{\left(\frac{\epsilon_y dx}{h/2}\right)^2}{\left(\frac{\epsilon_p dx}{h/2}\right)^2} \quad (A6)$$

Now using the results of equations (A4) and (A5) we have

$$\frac{M_p}{M_y} = \frac{3}{2} - \frac{1}{2} \frac{(d\alpha_y)^2}{(d\alpha)^2} \quad (A7)$$

With the view of attempting to make equation (A7) more general so that it can be used for beam cross sections other than rectangular, Seely and Smith substitute K for 3/2 and n = K - 1 for 1/2. K now represents a geometrical shape factor equal to  $M_{fp}/M_y$ , i.e., the ratio of the fully plastic moment to the yield moment. Table 6.1 of Faupel's book<sup>3</sup> gives values of K for different beam cross sections. Substituting these parameters, we have

$$\frac{M_p}{M_y} = K - n \left( \frac{d\alpha_y}{d\alpha} \right)^2 \quad (A8)$$

---

<sup>3</sup>J. H. Faupel, Engineering Design, Wiley and Sons, 1964, p. 395.

Solving for  $d\alpha$

$$K - \frac{M_p}{M_y} = n \left( \frac{d\alpha_y}{d\alpha} \right)^2$$

$$d\alpha^2 = n \frac{d\alpha_y^2}{K - \frac{M_p}{M_y}}$$

$$d\alpha = \frac{n^{1/2} d\alpha_y}{\sqrt{K - \frac{M_p}{M_y}}} \quad (A9)$$

Using Hooke's law and the flexure formula and substituting for  $\epsilon$  in (B4) we have

$$d\alpha_y = \frac{M_y}{EI} dx$$

Substituting this into equation (A9) we obtain the desired expression:

$$d\alpha = \frac{M_y}{EI} \frac{n^{1/2} dx}{\sqrt{K - \frac{M_p}{M_y}}} \quad (A10)$$

## APPENDIX B

At incipient yielding on the outside fiber we can determine an expression for  $d\alpha$  as follows. By referring to Figure B1 which indicates that planes remain plane and further assuming that the tangent of the angle equals the angle for small deformations, we have

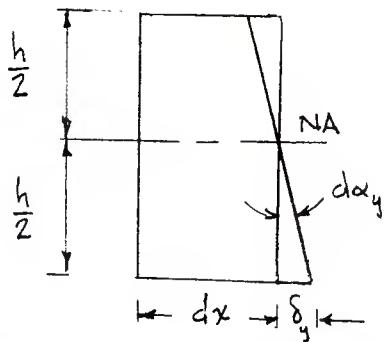


Figure B1. Strain vs. Depth of Cross-Section.

$$\tan(d\alpha_y) = d\alpha_y = \frac{\delta_y}{h/2} \quad (B1)$$

From an expression for engineering strain, we obtain

$$\epsilon_y = \frac{(dx + \delta_y) - dx}{dx} = \frac{\delta_y}{dx} \quad (B2)$$

hence

substituting into equation (A1), we get

$$d\alpha_y = \frac{\epsilon dx}{h/2} \quad (B5)$$

TECHNICAL REPORT INTERNAL DISTRIBUTION LIST

	<u>NO. OF COPIES</u>
COMMANDER	1
CHIEF, DEVELOPMENT ENGINEERING BRANCH	1
ATTN: DRDAR-LCB-DA	1
-DM	1
-DP	1
-DR	1
-DS	1
-DC	1
CHIEF, ENGINEERING SUPPORT BRANCH	1
ATTN: DRDAR-LCB-SE	1
-SA	1
CHIEF, RESEARCH BRANCH	2
ATTN: DRDAR-LCB-RA	1
-RC	1
-RM	1
-RP	1
CHIEF, LWC MORTAR SYS. OFC.	1
ATTN: DRDAR-LCB-M	
CHIEF, IMP. 81MM MORTAR OFC.	1
ATTN: DRDAR-LCB-I	
TECHNICAL LIBRARY	5
ATTN: DRDAR-LCB-TL	
TECHNICAL PUBLICATIONS & EDITING UNIT	2
ATTN: DRDAR-LCB-TL	
DIRECTOR, OPERATIONS DIRECTORATE	1
DIRECTOR, PROCUREMENT DIRECTORATE	1
DIRECTOR, PRODUCT ASSURANCE DIRECTORATE	1

NOTE: PLEASE NOTIFY ASSOC. DIRECTOR, BENET WEAPONS LABORATORY, ATTN: DRDAR-LCB-TL, OF ANY REQUIRED CHANGES.

TECHNICAL REPORT EXTERNAL DISTRIBUTION LIST (CONT.)

	<u>NO. OF COPIES</u>		<u>NO. OF COPIES</u>
COMMANDER US ARMY RESEARCH OFFICE P.O. BOX 12211 RESEARCH TRIANGLE PARK, NC 27709	1	COMMANDER DEFENSE TECHNICAL INFO CENTER ATTN: DTIA-TCA CAMERON STATION ALEXANDRIA, VA 22314	12
COMMANDER US ARMY HARVEY DIAMOND LAB ATTN: TECH LIB 2800 POWDER MILL ROAD ADELPHIA, MD 20783	1	METALS & CERAMICS INFO CEN BATTELLE COLUMBUS LAB 505 KING AVE COLUMBUS, OHIO 43201	1
DIRECTOR US ARMY INDUSTRIAL BASE ENG ACT ATTN: DRXPE-MT ROCK ISLAND, IL 61201	1	MECHANICAL PROPERTIES DATA CTR BATTELLE COLUMBUS LAB 505 KING AVE COLUMBUS, OHIO 43201	1
CHIEF, MATERIALS BRANCH US ARMY R&S GROUP, EUR BOX 65, FPO N.Y. 09510	1	MATERIEL SYSTEMS ANALYSIS ACTV ATTN: DRXSY-MP ABERDEEN PROVING GROUND MARYLAND 21005	1
COMMANDER NAVAL SURFACE WEAPONS CEN ATTN: CHIEF, MAT SCIENCE DIV DAHLGREN, VA 22448	1		
DIRECTOR US NAVAL RESEARCH LAB ATTN: DIR, MECH DIV CODE 26-27 (DOC LIB) WASHINGTON, D. C. 20375	1		
NASA SCIENTIFIC & TECH INFO FAC. P. O. BOX 8757, ATTN: ACQ BR BALTIMORE/WASHINGTON INTL AIRPORT MARYLAND 21240	1		

NOTE: PLEASE NOTIFY COMMANDER, ARRADCOM, ATTN: BENET WEAPONS LABORATORY,  
DRDAR-LCB-TL, WATERVLIET ARSENAL, WATERVLIET, N.Y. 12189, OF ANY  
REQUIRED CHANGES.

TECHNICAL REPORT EXTERNAL DISTRIBUTION LIST

<u>NO. OF COPIES</u>	<u>NO. OF COPIES</u>		
ASST SEC OF THE ARMY RESEARCH & DEVELOPMENT ATTN: DEP FOR SCI & TECH THE PENTAGON WASHINGTON, D.C. 20315	1	COMMANDER US ARMY TANK-AUTMV R&D COMD ATTN: TECH LIB - DRDTA-UL MAT LAB - DRDTA-RK WARREN, MICHIGAN 48090	1
COMMANDER US ARMY MAT DEV & READ. COMD ATTN: DRCDE 5001 EISENHOWER AVE ALEXANDRIA, VA 22333	1	COMMANDER US MILITARY ACADEMY ATTN: CHMN, MECH ENGR DEPT WEST POINT, NY 10996	1
COMMANDER US ARMY ARRADCOM ATTN: DRDAR-LC -LCA (PLASTICS TECH EVAL CEN) -LCE -LCM -LCS -LCW -TSS (STINFO) DOVER, NJ 07801	1	US ARMY MISSILE COMD REDSTONE SCIENTIFIC INFO CEN ATTN: DOCUMENTS SECT, BLDG 4484 REDSTONE ARSENAL, AL 35898	2
	1	COMMANDER REDSTONE ARSENAL ATTN: DRSMI-RRS -RSM ALABAMA 35809	1
	2	COMMANDER ROCK ISLAND ARSENAL ATTN: SARRI-ENM (MAT SCI DIV) ROCK ISLAND, IL 61202	1
	1	COMMANDER HQ, US ARMY AVN SCH ATTN: OFC OF THE LIBRARIAN FT RUCKER, ALABAMA 36362	1
DIRECTOR US ARMY BALLISTIC RESEARCH LABORATORY ATTN: DRDAR-TSB-S (STINFO) ABERDEEN PROVING GROUND, MD 21005	1	COMMANDER US ARMY FGN SCIENCE & TECH CEN ATTN: DRXST-SD 220 7TH STREET, N.E. CHARLOTTESVILLE, VA 22901	1
COMMANDER US ARMY ELECTRONICS COMD ATTN: TECH LIB FT MONMOUTH, NJ 07703	1	COMMANDER US ARMY MATERIALS & MECHANICS RESEARCH CENTER ATTN: TECH LIB - DRXMR-PL WATERTOWN, MASS 02172	2
COMMANDER US ARMY MOBILITY EQUIP R&D COMD ATTN: TECH LIB FT BELVOIR, VA 22060	1		

NOTE: PLEASE NOTIFY COMMANDER, ARRADCOM, ATTN: BENET WEAPONS LABORATORY, DRDAR-LCB-TL, WATERVLIET ARSENAL, WATERVLIET, N.Y. 12189, OF ANY REQUIRED CHANGES.